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ARL-TR-1800

September 1998

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Army Research Laboratory
Aberdeen Proving Ground, MD 21005-5067

ARL-TR-1800 **September 1998**

The Feasibility of Vibration Diagnostics Measurements in a Gas Turbine Engine

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Abstract

A variety of analytical and experimental techniques has been explored to determine the efficacy and viability of application for diagnostics of a tank gas turbine engine. Engine performance criteria, along with mechanical maintenance programs, have been identified to be used in the turbine engine diagnostics program. Fast-Fourier analysis, Hilbert transforms, wavelets, and Kurtosis techniques were explored as the governing tools in determining engine functionalities. A proof-in-principle test was performed via spectrum analysis that clearly illustrated that a signal-rich environment, as provided by the measurements performed in this specific study, is required for engine performance analysis and diagnostics.

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1. Introduction

A diagnostic system should be developed that demonstrates the application of the sonic concept and analysis techniques to monitoring the mechanical degradation of turbine engines and associated power-train systems. Inputs from a number of sensors should be normalized to the level of a predetermined band of noise that is characteristic of a given engine, thereby providing a reference base for component condition limits and analyzer system calibration.

Besides the data input, one or more of the sensors should monitor the engine rpm by sending a discrete signal produced by an engine component into a phase-locked filter. The output of the phase-locked filter should then be processed by a frequency-ratio generator. The resulting engine rpm tracking function enables precise control of the narrow-band center frequency placement.

The location and number of microphones required for complete signal coverage of the sound spectrum should be determined during a preliminary data-acquisition survey. This survey should also be performed to determine the component signal(s) to be utilized for rpm tracking and to establish basic component limits for the diagnostic program. Further refinement of the diagnostic programs is accomplished during training and implementation of the system.

A real-time spectrum analyzer (RTA) generates the spectrum analysis of the vibration data instantaneously and in an easily comprehensible form. Also existing are other techniques for data acquisition, such as a signature-ratio analysis, a technique for adding clarity and continuity to rotating machinery data; spectrum averaging, which improves the signal-to-noise ratio in the analysis of noisy signals; and spectrum translation, which allows extremely fine resolution. The data management task can be greatly facilitated by using the real-time analyzer in conjunction with a small digital computer. A microcomputer can be used to store baseline information from which behavioral trends of a machine can be monitored.

Cataloging of vibration and noise spectra taken at regular intervals on a piece of equipment can be extremely valuable, both in diagnosing potential problems and in predicting machine life. There are two approaches that can be taken in this type of program. Firstly, when a large number of the pieces of equipment are available for testing, vibration data can be taken for a batch of these components. From this data, a mean criterion for vibration spectra can then be established. Individual items may then be tested against this mean value and decisions made on a "Go/No-Go" basis. Secondly, as in the case of a gas turbine or other large machines, vibration data may be taken at the beginning of the machine's service life and at periodic intervals thereafter. Every machine will exhibit changes in its vibration spectra due to gradual wear. By monitoring the vibration on a regular basis, any irregular changes in vibration level can be spotted before they get to a point that could cause actual machine damage.

The overall vibration monitors are dedicated sentries working full time and are very much a necessary part of maintaining safe machinery operation. In fact, one very worthwhile program for implementing the use of a spectrum analyzer is in response to a first-stage alarm given by such overall monitors; then, by exploring the spectrum, accurate judgements can be made and, perhaps, emergency shutdown prevented. As an example, perhaps a decision can be made to decrease the load or speed of a machine by 10%, thereby extending its operation through an important production period until repair or replacement can be scheduled. The basic limitation in using a swept-frequency narrow-band filter for analysis is the length of time required for the analysis. The narrower the filter, the more slowly it must be swept through the analysis frequency range.

RTA accomplishes an analysis in a short period of time, primarily due to the technique of digital sampling and time compression that speeds up the input data signal. This signal is translated to frequencies in the megahertz range. Analysis of this proportionately higher frequency can be done rapidly using a wide-band filter. Relative to the input-signal frequency range, this filter is narrow.

An averager is a very important part of the RTA system. In cases where the vibration signal contains noise, the important vibration components can be buried or concealed by the noise. The

averaging process allows the randomly occurring signals to approach a mean value, clearly revealing the important spectrum data.

A frequency translator allows for an ultranarrow-band filter to be used in the higher frequency range. This capability is important if vibration components are close in frequency. A typical example of this is where side bands are generated due to an amplitude-modulation process (Fieldhouse 1970). RTA can "zoom in" to separate and identify discrete signals when used with a translator.

One problem associated with vibration spectrum analysis can exist if rpm fluctuates slightly during the course of measurements because of inherent speed-control limitations or, perhaps, because of fouling. When this occurs, different signature-analysis records of a given machine will show vibration components occurring at different frequencies in the spectrum. A slight change in operating speed will produce an increasingly larger change at higher multiple-order frequencies. A solution to this problem lies in the use of a signature-ratio adapter with RTA.

The signature-ratio adapter records data directly in vibration amplitude vs. ratio (of component to fundamental speed) rather than vibration amplitude vs. frequency. In essence, it divides the frequency axis of the spectrum plot by rotational frequency of the machine. The harmonic relationships are automatically maintained at the same value(s) regardless of speed changes that occur between the data-taking periods. The signature-ratio concept also allows for order-related components to be more easily compared when looking at records produced at two different running speeds without having to do any mathematical gymnastics.

Any small speed variations occurring during the course of analysis, while supposedly operating at fixed speed, are automatically compensated for by using this technique. Thus, analyses performed in terms of signature ratio may be directly related to one another without mathematical correction. More importantly, suppose that spectrum averaging is used to deemphasize the background noise and permit critical higher order vibration components to be visible. Without signature ratio, any slight speed change that occurs during the averaging process will cause the higher order components to

change frequency and a smearing effect will result. Averaging using the signature-ratio concept can be accomplished without concern over minor speed variations.

The computer can do the otherwise manual task of predicting trends or of diagnosing machinery characteristics with a high degree of accuracy and at a very rapid rate. This can be done by giving the computer a machine's baseline performance spectrum data along with data gathered at regular intervals. The baseline information is gathered during installation of a new machine or following the overhaul of existing machinery. In the case of a machine that has been running for an extended period, useful baseline information can be established, as long as performance and vibration levels have been steady.

The second role in which a computer is useful is in a completely automated system. Many such systems are currently in use providing early warning of incipient problems or failures. These systems generally monitor temperatures, pressures, flows, and overall vibration data. They provide different levels of warning based on trends that are observed from these inputs (Gas Turbine World 1971). Some systems even have diagnostic capabilities. Few of these systems use vibration spectrum data as inputs. This is because vibration spectrum information prior to the advent of the real-time analyzer was impractical and costly. Today, the relatively low cost of the RTA makes this concept economically possible. Monitoring the vibration spectrum data in lieu of the overall vibration data yields a clearer picture of the machine's condition.

Displacement transducers, velocity transducers, accelerometers, microphones, and strain gauges each provide vibration and noise information useful for this type of data acquisition. Each transducer has specific operational characteristics that would limit a good data-acquisition program if only one type of transducer were used.

The displacement transducer, specifically the noncontacting probe, is a unique transducer in that it can measure the relative motion between two surfaces. It is the only transducer that can provide this type of information. Velocity and acceleration transducers measure forces that are relative to the earth's gravitational force. The displacement transducer has a good linear amplitude range but is

is limited to operation in the lower frequency ranges. Nicks and scratches on a rotating surface directly affect the output signal and spectrum characteristics when using a displacement transducer.

The velocity transducer operates in a frequency range that overlaps that of the displacement transducer. The velocity transducer should not be used at low frequencies where its amplitude response becomes nonlinear due to resonances. It has a limit in its upper operating frequency range, but it extends beyond that of the displacement transducer. By applying the velocity transducer output to an electronic signal integrator, a displacement level can be read. The velocity transducer is useful to determine vibration that exists at higher orders of rotational speed where a displacement transducer cannot respond. Many structural resonances occur in this frequency range (i.e., greater than 500 MHz). The velocity transducer, therefore, is useful to determine the kinematics of a machine's behavior.

There are many characteristics of a machine where extremely high-frequency information exists. For example, blade-passage frequencies can exist at 100 times running speed. Gear mesh frequencies and, in some cases, bearing frequencies can produce vibration information at these higher frequencies also. For this type of data, an accelerometer is required due to its extremely high operating frequency characteristics. The output of an accelerometer is calibrated in units of gravitational forces called "g's." Acceleration is the second derivative of displacement, which means that if its signal is passed through an electronic double integrator, the output can also be read in displacement units. The formula $g = 0.0511f^2x$ (where x is peak-to-peak displacement and f is frequency in revolutions/second) is derived from this relationship and provides a "connection" between the two units of measurement. Another advantage in using an accelerometer is the fact that it is relatively light and may not add appreciable damping to the structure.

In many cases, a microphone is useful for surveying running machinery. The narrow-band acoustic spectrum produced by the RTA allows accurate identification of the frequency at which noise exists. Knowing the precise frequency information, the source of the noise or vibration can be identified. Where the microphone allows for taking a "broad-view survey" to pinpoint a possible problem, it does not lend itself to repeatability. Since it is not a directional transducer, the

transmission paths for the noise cannot be defined. Many reflected transmission paths can be severely influenced by surroundings that might change from one time to another.

The strain gauge cannot be overlooked as a valuable transducer when a study is being made of the mechanical integrity of a machine. Spectrum data from the strain gauge can be as important to the design engineer as the value of strain itself. Often a strain gauge can be placed in areas on a structure where it is impossible to use other types of transducers due to a lack of space.

Much information stands to be gained by simply using a spectrum analyzer when a problem is suspected. In this case, it may not be necessary to compare to the spectrum with previously obtained data. Many excellent references are available that give guidelines for correlating basic machinery problems with their vibration spectra (Shuey 1972; Tustin 1971; Miller 1967). The RTA has the capability to provide the spectrum information, instantaneously revealing a machine's characteristics. A complete listing of all possible frequencies at which vibration could exist should be compiled and kept with the machine's portfolio. Such a listing should include information like running speed, types of bearings, the number of elements and size, gear-mesh frequencies, the number of turbine blades, speeds of accessory components, and information about structural resonances or critical frequencies. This would provide a rapid means by which the spectrum data could be resolved and where frequency components where high vibration levels exist could be identified. Soon after, a portfolio of case histories and vibration trend data would be compiled and used to make accurate predictions concerning the machine's operational characteristics and life expectancy.

A dedicated program of spectrum analysis would certainly be more fruitful. The spectrum of a signal from a vibration transducer produced by a running machine can be used as the "signature" or fingerprint of that particular machine's operational condition (Bannister and Donato 1971). The signature data should be carefully labeled with a date, pertinent operating conditions, transducer type, and location information and filed in a folder assigned to the particular machine.

A program of periodic spectrum analysis would provide new signatures to be compared with the baseline information. Any rapid changes with time in the signature could be carefully observed by

doing signature analysis more frequently to keep track of any trends that might exist. Normal wear over long operating periods should show a gradual trend that would be reflected in the analyses. If a particular frequency component begins to change at an increasing rate, it should be looked upon as an indicator of impending problems. This new trend in the spectrum data can be compared with trends in temperature and other data to help identify the nature of the problem. Accordingly, an orderly shutdown may be scheduled.

The meshing of gears can yield an abundance of information regarding age, wear, general condition, maintenance, strength, toughness, and diagnostics of gearing faults that are readily amenable to rigorous analysis. Among the many techniques available is signal-processing analysis which is especially suitable for pattern recognition, fault detection, and even the prognostics of engines as illustrated by Helfman, Dumer, and Hanratty (1995). Most, if not all, of the techniques that are commonly used necessitate a synchronous time averaging with respect to the gear pair analyzed so that extraneous disturbances can be adequately eliminated from the actual signal (Braun and Seth 1979; Brennan, Chen, and Reynolds 1997). This particular averaging involves sampling the vibration signal at a frequency that is synchronized with the rotation of a particular gear under examination with subsequent calculation of the ensemble average of the signal over many revolutions. Kurtosis employs the computation of the fourth statistical moment of the time history of the vibration signal as done by Stewart (1977). A normalization gives a single dimensionless number that indicates faulty gears. It was also shown that normal gears generate a reasonably uniform pattern of the tooth-meshing frequency and, hence, the Kurtosis value should be relatively low; whereas, if a tooth is damaged in any way, the tooth meshing becomes modulated, thereby raising the Kurtosis value, as demonstrated by Randall (1982).

McFadden and Smith (1985) examined the analytic envelope function for detecting gear faults in order to look at information concerning the amplitude and phase modulation of the gear meshing. The function in question can be determined by band-pass filtering the time-averaged signal about one of the dominant meshing harmonics and then performing a Hilbert transform. McFadden (1986) used this particular transform to detect a crack in a spear gear; he also found that phase or frequency modulation of the vibration was a more important indicator than amplitude modulation. Staszewski

and Tomlinson (1994) used wavelet transforms to detect gearing faults. The technique of wavelet transforms generates a three-dimensional complex time-frequency graph of the gear vibration that can be somewhat difficult to interpret; but, its variable time and frequency resolution enable the evolution of the spectrum of a signal with time to be analyzed with more flexibility than is possible with established Fourier-transform-based methods (Braun and Seth 1979; Brennan, Chen, and Reynolds 1997).

2. Methodology

According to Mark (1978), the principal source of vibratory excitation of a pair of involute gears is the unsteady component of the relative angular motion of the meshing gears, which is caused by the compliance of the gear teeth and manufacturing inaccuracies. If we consider a pair of gears, one of which has T teeth and is rotating with frequency of f_r in revolutions/second, which mesh under a constant load and speed, then the fundamental meshing vibration can be given as $f_m = Tf_r$ [Hz]. The meshing vibration can then be expressed as a sum of N + 1 dominant harmonics, each of amplitude X_n such that

$$X(t) = \sum_{n=0}^{N} X_n \cos(2\pi n f_m t + \phi_n), \tag{1}$$

where ϕ_n is the phase angle of the nth harmonic. If the gear contains a localized defect like a partial tooth failure, then the amplitude and frequency of the vibration of the affected gear will be modulated according to Randall (1982). The modulated gear-meshing vibration y(t) was given by McFadden (1986) as

$$y(t) = \sum_{n=0}^{N} X_n [1 + a_n(t)] \cos(2\pi n f_m t + \phi_n + b_n(t)), \tag{2}$$

where $a_n(t)$ and $b_n(t)$ are the amplitude- and phase-modulating functions, respectively. These can be written as a sum of harmonics of the gear rotational frequency such as

$$a_n(t) = \sum_{p=0}^{Q} A_{np} \cos(2\pi p f_r t + \phi_n \alpha_{np}), \tag{3}$$

and

$$b_n(t) = \sum_{s=0}^{S} A_{ns} \cos(2\pi s f_r t + \beta_{ns}). \tag{4}$$

It is well known that amplitude modulation produces side bands around the carrier frequencies, which happen to be the gear-meshing frequency and its harmonics. These side bands are usually spaced at intervals of the modulating-function frequencies. If per chance the modulating function contains only one frequency (e.g., the frequency of two teeth in a set of gears touching each other), then there may be only a single pair of side bands (e.g., the result of periodic variations in tooth loading caused by eccentricity of gears, tooth profile errors, or irregular wear). If the modulation function consists of many frequencies caused by a local tooth defect, then many side bands could be generated (Brennan, Chen, and Reynolds 1997).

Even if the modulating function is a single one, frequency modulation can possibly produce a family of side bands that could be spaced at intervals of the modulating frequency. The bandwidth of the modulating frequency function is usually dependent on the frequency deviation of the carrier signal. In gearing systems such as in the gear box of a gas turbine engine, there exist both an amplitude modulation and phase modulation such that the resulting spectrum will contain side bands of the corresponding function. Although it has been determined that amplitude and frequency modulation produce symmetrical families of side bands when acting alone, the phase relationships on either side of the carrier frequency are usually different and the combination of the two families of side bands can reinforce or cancel each other, producing an asymmetrical family of side bands.

Kurtosis analysis involves the fourth statistical moment about the mean of a random signal (variable) (Brennan, Chen, and Reynolds 1997). This gives an indication of the peakedness of the signal but is resilient to rare extreme data points or anomalies. Since the mean, or the first moment about zero, of a random variable X can be given by $\mu = E(X)$, where E is the expectation operator, and the variance or the second moment can be readily defined by $\mu_2 = \sigma^2 = E\{(X - \mu)^2\}$, then higher order moments can be defined in a similar fashion: $\mu_k = E\{(X - \mu)^k\}$, where k = 3, 4, 5, . . . , Kurtosis is simply the fourth moment normalized to the square of the variance:

$$\text{Kurtosis} = \frac{\mu_4}{\sigma^4}. \tag{5}$$

Equation (5) can be used to calculate the Kurtosis of the signal when the vibration signal has been time-averaged to remove frequency components not related to the gears of interest. If we are dealing with a normal distribution (i.e., a Gaussian signal), the Kurtosis turns out to be 3. As the peakedness of the vibration signal increases, as is the case for a defective gear tooth, then Kurtosis increases.

McFadden (1986) described a Hilbert-transform-based demodulation process in which the given signal must be bandpass-filtered around one of the dominant harmonics including all of its side bands. The reason for the filtering is because it is not possible for the Hilbert transform to demodulate the envelope and phase of the whole vibration signal. For the *n*th harmonic, the signal can be described as

$$Z_n(t) = X_n[1 + a_n(t)]\cos[2\pi n f_m t + \phi_n + b_n(t)]. \tag{6}$$

$Z_n(t)$ can then be combined with its Hilbert transform to yield the analytic function $C_n(t)$, viz.,

$$C_n(t) = Z_n(t) - j\,H\,[Z_n(t)], \tag{7}$$

where H denotes a Hilbert transform and j is simply the square root of -1. If we substitute equation (6) into (7) and then perform the Hilbert transform, we are left with the analytic function in its expanded form (McFadden 1986):

$$C_n(t) = X_n[1 + a_n(t)]e^{j[2\pi n f_m t + \phi_n + b_n(t)]}. \tag{8}$$

Clearly, the analytic function is complex where its modulus is the envelope of the modulated carrier signal and is a function of the amplitude modulation $a_n(t)$, and the phase is a function of phase or frequency modulation, $b_n(t)$. Hence, if we were to plot the modulus and phase of the analytic function, then gear defect modulating the meshing vibration might be able to be identified without too much difficulty. The problem with this type of demodulation technique lies in the process of bandpass-filtering a dominant harmonic and all its side bands (i.e., if many side bands are generated, then it is possible that there will be interference between adjacent harmonics of the gear-meshing frequency, and, hence, it may not be clear exactly how to choose the cutoff frequency of the bandpass filter). A filter with a narrow-pass band will more than likely miss some of the higher order side bands of the chosen harmonic, whereas as a filter with a wider pass band will more than likely pass some of the side bands from adjacent harmonics (Brennan, Chen, and Reynolds 1997). Hence, in either case, the analytic function will be distorted and will not give a true picture of the actual amplitude and phase modulation of the gear under scrutiny. This has been well documented by Reynolds (1995).

The wavelet-transform conceptualization can be most easily understood through the short time Fourier transform, which is a transform allowing the evolution with time of the spectrum of any given signal as generated by the Fourier transform itself. The short-time transform is necessary when attempting to detect a defective gear due to the nonstationary nature of the vibrational signal. The short-time Fourier transform (STFT) can be cast as (Bentley and McDonnell 1994):

$$\mathrm{STFT}(\tau, f) = \int_{-\infty}^{+\infty} y(t) g(t-\tau) e^{-j2\pi f t} dt, \tag{9}$$

where y(t) is the vibrational signal given in equation (2) and $g(t-\tau)$ is a windowing function around time τ. The actual window function is shifted in time over the entire signal, and consecutive overlapped transforms are then performed, which gives a complete (i.e., from signal incipiency to cutoff point) description of the evolution of the spectrum. A frequency-time description of the signal can be plotted if these transforms can be arranged in chronological order. The window function g(t) allows the generation of a frequency-time description of the signal; however, it is noted to have a harmful effect upon the frequency resolution itself. More frequently, a short time window yields good time resolution Δt results but poor frequency resolution Δf; likewise, a long time window yields poor time resolution results yet good frequency resolution, and according to the Heisenberg uncertainty principle as noted in (Bentley and McDonnell 1994),

$$\Delta t \Delta f \geq \frac{1}{4\pi}. \tag{10}$$

It is also well known that, when a Gaussian window function is employed, the resulting relationship reverts to an equality and the STFT is commonly referred to as a Gabor transform (Onsay 1994). A typical problem with the STFT is that the shape and length of the time window that is responsible for the time and frequency resolution do not change throughout the given analysis period which ultimately causes the time and frequency resolutions to also not change. According to a treatise by Onsay (1994), short time windows with high frequencies will yield good frequency resolution; this phenomena is clearly evidenced by the wavelet facility since the frequency scale is greatly expanded over appreciable time intervals. In addition, although the STFT had the interpretation of a Fourier transform of the window signal $y(t)g(t-\tau)$, we can also describe this particular function as the decomposition of the signal y(t) into the windowed basis function

$$q_{\tau,f}(t) = g(t-\tau)e^{j2\pi ft}. \tag{11}$$

By basis function, we mean a complete set of functions that can be combined as a weighted sum in order to construct a given signal. For the case of the STFT, the basis functions will be the sine and

cosine functions windowed by the function g(t) and centered around time τ. Hence, we can now formulate a general equation for the STFT as the inner product of the signal and basis functions, viz.:

$$\mathrm{STFT}(\tau,f) = \int_{-\infty}^{+\infty} y(t)\overline{q_{\tau,f}}(t)dt. \qquad (12)$$

In addition, the wavelet transform can be expressed in terms of its basis functions commonly referred to as wavelets, by invoking equation (12) such that the frequency variable is replaced by the scale variable *a*, and the time-shift variable τ is replaced by b. The wavelet or basis functions can be expressed as

$$h_{a,b}(t) = \frac{1}{\sqrt{a}} h^* \frac{t-b}{a} dt. \qquad (13)$$

where * denotes the complex conjugate. If we now substitute equation (13) into equation (12), then we are left with the continuous wavelet transform (CWT)

$$\mathrm{CWT}(a,b) = \frac{1}{\sqrt{a}} \int_{-\infty}^{+\infty} y(t)\ h^* \frac{t-b}{a} dt. \qquad (14)$$

Here, we can readily see that the wavelet transform performs a decomposition of the signal y(t) into a weighted set of scaled wavelet functions. However, since the wavelet transform accommodates variable window lengths, there is variable resolution and, thus, an increase in performance can be realized notwithstanding that the wavelet transform does not overcome the uncertainty principle. Hence, we are led to conclude that the time window decreases in length as the frequency increases chiefly due to the scale factor a in the time window function per se. Therefore, at low frequencies where the time window is large, there may be poor resolution but with good frequency resolution; likewise, at high frequencies, there is good time resolution but poor frequency resolution.

The bandwidth of a wavelet is proportional to its center frequency so that the wavelet transform acts like a bank of constant relative bandwidth filters that give a logarithmic coverage of the frequency domain (Brennan, Chen, and Reynolds 1997). It is this flexible scheme of time and frequency localization that makes the wavelet transform attractive for the analysis of signals involving discontinuities or transients. Many wavelets are available as basis functions; however, the question of which wavelet is suited to a particular application is still being investigated. For example, if we are to consider the detection of defective teeth in a gear mesh, then we might want to consider the Morlet wavelet (Morlet 1982) since it closely resembles the Fourier transform that is very familiar for most people since it is an analytic sinusoid within a Gaussian envelope. Furthermore, it has been found (Wang and McFadden 1995) that nonorthogonal wavelets gave a slightly better indication of faults. The difference between the Morlet and the nonorthogonal wavelet is in the basis function, i.e., the Morlet wavelet can be expressed as

$$h(t) = e^{\frac{-t^2}{2}} e^{j2\pi f_0 t}. \tag{15}$$

and the nonorthogonal wavelet can be expressed as

$$h(t) = ce^{-\sigma^2 t^2} e^{j2\pi f_0 t}. \tag{16}$$

3. Conclusions

Spectrum and sonic analysis techniques have been developed and successfully applied as maintenance tools used for monitoring the mechanical condition of rotating components in gas turbine engines and drive systems operating in aircraft and test cell installations as evidenced in Curtiss-Wright Corporation (1968), Frarey and Handjani (1966), Frarey and Zabriskie (1965), and Mechanical Failures Prevention Group (1971). The concept has been applied to determine the mechanical integrity of industrial gas turbine installations and to monitor the dynamic operating characteristics of transmissions during development testing.

Diagnostic monitoring of a transmission system achieved excellent correlation between acoustic and spectrometric oil analysis techniques in the case of a spalled roller on the left nacelle pinion bearing (Curtiss-Wright Corporation 1968; Frarey and Handjani 1966; Frarey and Zabriskie 1965; Mechanical Failures Prevention Group 1971).

Spectrum and sonic analysis techniques have been implemented as standard overhaul maintenance procedures to assure a higher degree of quality control during postoverhaul testing of customer gas turbines. This added capability has also reduced maintenance costs related to establishing engine acceptance by defining the level of maintenance required for corrective action as indicated by Braun (1986).

Use during production overhaul testing has demonstrated the capability to predict the amount of unbalance present in compressor rotors from the value of the side band amplitude ratios. Engines rejected for excessive vibration, but found to be mechanically sound by the sonic engine analyzer, have been further investigated for faulty vibration transducers to prevent possible rejection back to overhaul as indicated by Reynolds (1985).

Data have been recorded and analyzed for uprated power-development testing of planetary gear boxes of double helical gear configurations. The signal characteristics detected in the acoustic spectrum were an indication of mechanical and dynamic changes occurring during the various testing phases. They were related to carrier eccentricity, unequal planetary gear loading, and degree of applied torque. These data, coupled with that obtained from other methods of vibration instrumentation, have been correlated with results derived from mathematical models to assist in defining a specific direction for design and testing. Acoustic patterns obtained during locked-torque testing have been related to differences in system fits and clearances. The degree of amplitude modulation of gear-meshing frequencies and detection of resonant frequencies at various operating conditions were monitored to assist in defining test profiles for the experimental assemblies being evaluated as noted by Wang and McFadden (1995).

As a trial run for a proof-in-concept principle, fast-Fourier transforms (FFT) were generated by the Turbine Engine Diagnostics Team of the U.S. Army Research Laboratory (ARL) from both a turbine startup condition and gearbox rundown of a tank turbine engine. Data were collected on the engine via an accelerometer placed originally on the turbine bearing shaft housing and then moved to the gearbox housing for the coast down. Since we did not have the luxury of a tachometer signal, much of the data were difficult to ascribe to each individual component; however, these data points/plots demonstrated that the two locations chosen were very signal-rich and perhaps may give us a good start in determining the proper course of technical action when funding and equipment become available. Pictured in Figure 1 is the frequency vs. time plot of data from the turbine startup. A "waterfall" data display can then be generated, as illustrated in Figure 2, which clearly shows some of the engine signal peaks paving the way for further research and investigation.

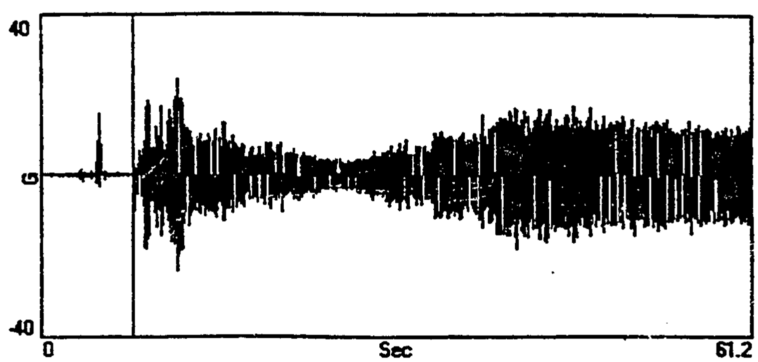

Figure 1. Time-Domain Distribution of Frequency vs. Time of a Rotating Center Shaft for a Gas Turbine Engine During Startup.

Selection of a couple of FFTs from Figure 2 were isolated to show how a signal can be decomposed via spectrum analyzer, as depicted in Figures 3 and 4. Figure 4, in particular, is from the steady-state speed data toward the end of the data file collected.

Figures 5 and 6 show the data taken from the gear box alone. This location also indicated a signal-rich environment for further research and investigation. Better locations certainly exist to take

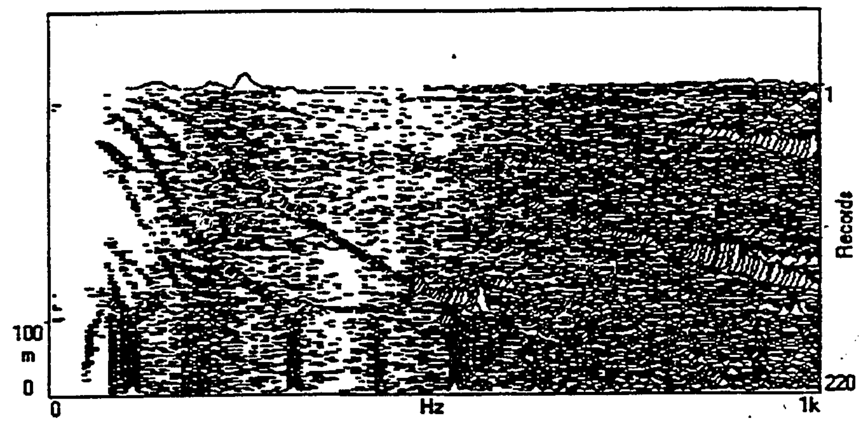

Figure 2. Time-Domain Distribution of Frequency vs. Time for a Rotating Center Shaft (Waterfall Display) For a Gas Turbine Engine.

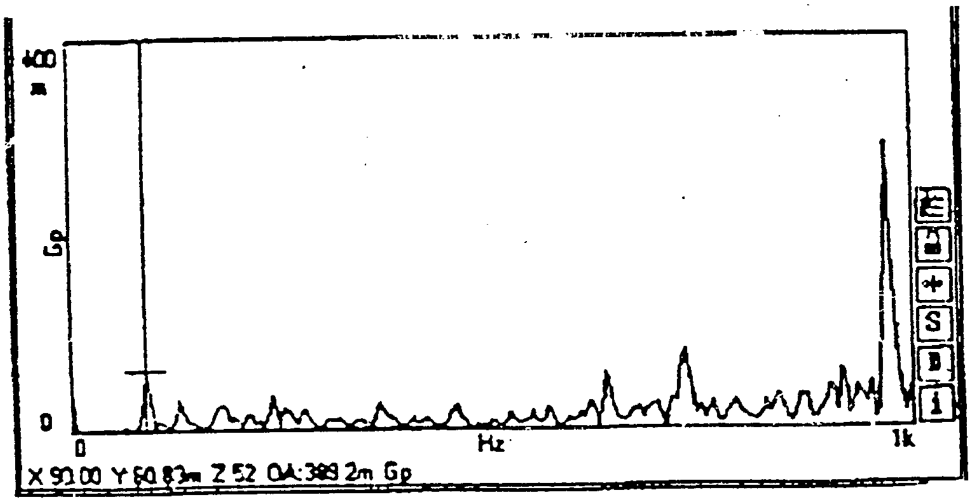

Figure 3. Decomposition of Time Distribution Signal of a Rotating Shaft for a Gas Turbine Engine.

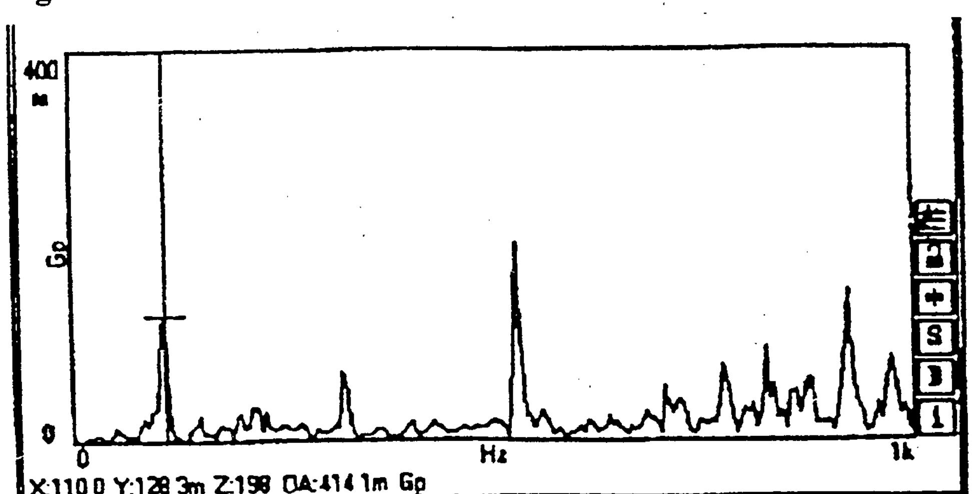

Figure 4. Decomposition of Time Distribution Signal of a Rotating Shaft During Startup for a Gas Turbine Engine.

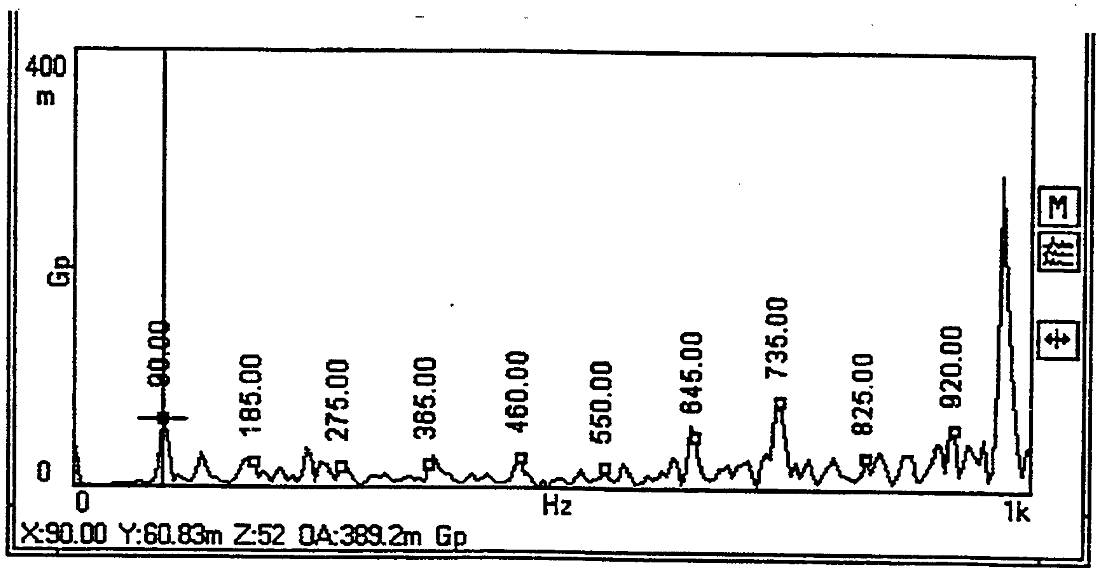

Figure 5. Amplitude vs. Frequency Signal of the Gear Box Signal of a Gas Turbine Engine.

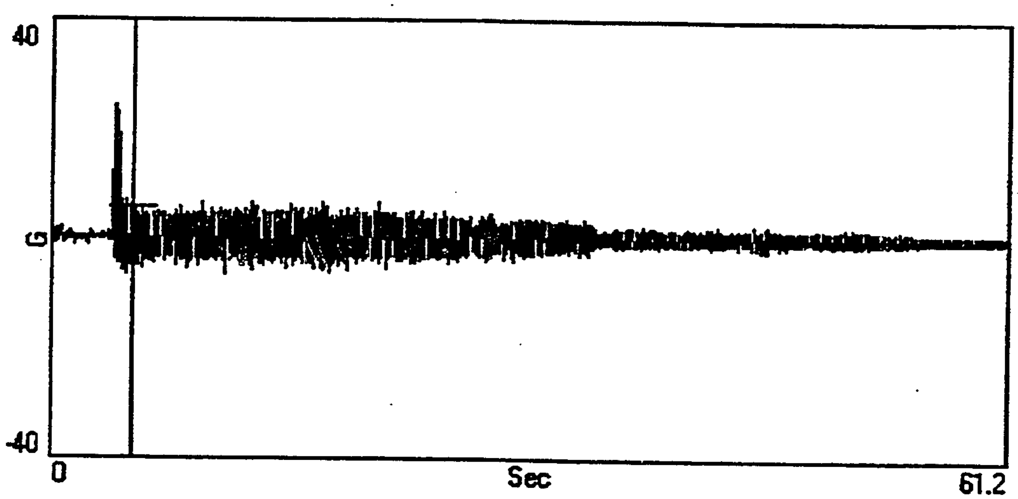

Figure 6. Frequency vs. Time Signal Distribution of the Gear Box of a Gas Turbine Engine.

data samples, but the locations thus chosen proved that signal-rich analyses can be achieved via spectrum analysis for the turbine engine to be used in engine diagnosis and performance.

Three potential techniques to diagnose mechanical faults have been discussed in the introduction. Although these aforementioned techniques showed some promise theoretically, the experimental results, although limited in nature, demonstrated that none of the techniques can provide an obvious

and reliable diagnosis without much more testing and analysis with the appropriate equipment. One school of thought entertains the idea that the test rig dynamics (testing equipment, etc.) could interfere with the actual signal of the individual mechanical component (i.e., the application of the uncertainty principle). This type of phenomena is likely to occur in any real-world monitoring system resulting in the amplification or attenuation of important excitation frequencies that carry pertinent information about the "normal" signature vs. the "abnormal" signature, which can be used for diagnostic purposes. One possible way that one might overcome this problem is to mount transducers directly on the gear(s) themselves or as close as practicable to the individual mechanical elements under scrutiny without any actual contact of the sensors with the excitational sources; however, this will significantly increase the complexity of a monitoring system and certainly requires much more extensive investigation. If we were to try to employ the demodulation technique, we would probably not be able to detect impulsive disturbances because of the problems in bandpass filtering the vibration signal around a dominant harmonic due to the interference of the side bands from adjacent harmonics caused by other vibratory sources within the signal environment. Demodulation would probably be good for detecting low-frequency modulation caused by some type of eccentricity, especially when the signal source is enriched with low-energy sources yielding lower spectrum harmonics as what might occur in varying gear-meshing ratios.

The wavelet transform could possibly be used in the detection of local gear tooth faults. Although the three-dimensional (3-D) magnitude and phase plots could be generated given the right software, these plots will probably be quite difficult to interpret, as described by the literature previously cited; however, it is well worth the time and effort to experiment with such software, especially if we just look at gears, although it is still not clear at the present time which is the best basis function to employ.

Kurtosis could possibly detect certain mechanical component defects that generate large impulsive signals; this type of technique will probably not help us with signals that are considerably regular in nature.

All techniques mentioned in this work have merits in the detection of faults—in particular, gearing faults. Kurtosis is a useful indicator of potential problems and is a relatively inexpensive way of crudely monitoring the state of certain mechanical systems inherent in the gas turbine engine. The techniques using Hilbert transforms and wavelet transforms are certainly much more sophisticated than simple time averaging and could be useful in the periodic checking of the actual health of a mechanical vibratory system or component in the engine. Much more work is required to determine the optimum positioning of sensors in conjunction with these techniques, and it is expected that more experimental research will follow when we perform extensive data acquisition and analysis of our turbine engine in the near future.

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REPORT DOCUMENTATION PAGE

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1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	September 1998	Final, January 1998 - July 1998

4. TITLE AND SUBTITLE

The Feasibility of Vibrational Diagnostics Measurements in a Gas Turbine Engine

5. FUNDING NUMBERS

4B010503-3500

6. AUTHOR(S)

T. A. Korjack

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

U.S. Army Research Laboratory
ATTN: AMSRL-IS-CI
Aberdeen Proving Ground, MD 21005-5067

8. PERFORMING ORGANIZATION REPORT NUMBER

ARL-TR-1800

9. SPONSORING/MONITORING AGENCY NAMES(S) AND ADDRESS(ES)

10.SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

A variety of analytical and experimental techniques has been explored to determine the efficacy and viability of application for diagnostics of a tank gas turbine engine. Engine performance criteria, along with mechanical maintenance programs, have been identified to be used in the turbine engine diagnostics program. Fast-Fourier analysis, Hilbert transforms, wavelets, and Kurtosis techniques were explored as the governing tools in determining engine functionalities. A proof-in-principle test was performed via spectrum analysis that clearly illustrated that a signal-rich environment, as provided by the measurements performed in this specific study, is required for engine performance analysis and diagnostics.

14. SUBJECT TERMS

vibration, diagnostics, gas turbine engine, fast-Fourier analysis, Hilbert transforms, wavelets, Kurtosis

15. NUMBER OF PAGES

28

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	UL

NSN 7540-01-280-5500

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